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HEAT EMISSION AND FRICTION HEADS OF HOT-WATER RADIATORS AND CONVECTORS

BY
FREDERICK E. GIESECKE
AND
ALONZO P. KRATZ

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BY

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HEAT EMISSION AND FRICTION HEADS OF HOT-WATER RADIATORS AND CONVECTORS

I. Introduction

1. *Objects of Investigation.*—The principal objects of this investigation were to measure the heat emission rates and the friction heads of hot-water radiators and convectors of various sizes and forms; and to determine the effect of the following factors on the heat emission rates and friction heads:

(1) The type and size of the unit
(2) The size of the pipe connections
(3) The type of the pipe connections
(4) The velocity of the water
(5) The temperature of the water.

A further object was to determine the difference, if any, between the rates of heat emission obtained when either steam or hot water at the same mean temperature was used as the heating medium in connection with the same radiators and convectors.

2. *Acknowledgments.*—The investigation was conducted as a part of the work of the Engineering Experiment Station and the Department of Mechanical Engineering of the University of Illinois in cooperation with the American Society of Heating and Ventilating Engineers and the Institute of Boiler and Radiator Manufacturers. The studies were carried on under the administrative sponsorship of Dean M. L. Enger, Director of the Engineering Experiment Station, and of Professor O. A. Leutwiler, Head of the Department of Mechanical Engineering.

Acknowledgment is hereby made to Professor M. K. Fahnestock, Research Professor of Mechanical Engineering, who was largely responsible for the design and construction of the test plant and who also assisted in planning the test program.

Acknowledgment is also made to Mr. Sam Sachs, formerly Research Assistant, for services rendered in connection with the design and construction of the testing plant, and to Mr. W. S. Harris, Research Associate, for assistance in the testing program.

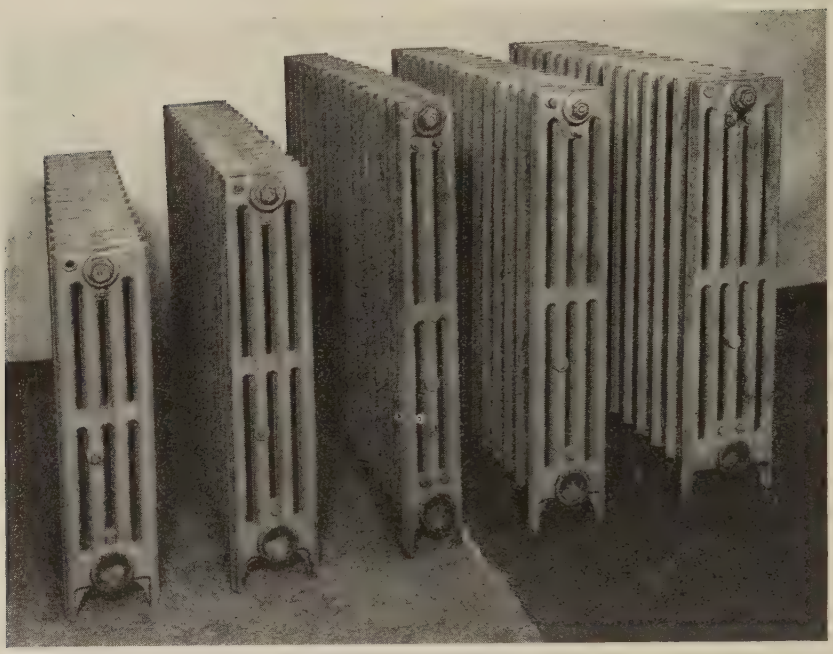

FIG. 1. SMALL-TUBE RADIATORS

II. DESCRIPTION OF RADIATORS AND CONVECTORS

3. *Radiators.*—The radiators tested were small-tube radiators of various heights and lengths and with different numbers of tubes per section. Five of the radiators are shown in Fig. 1.

For the studies on friction heads the units tested were

(1) One 19-in., 4-tube, 7-section cast-iron radiator

(2) One 19-in., 4-tube, 15-section cast-iron radiator

(3) One 19-in., 4-tube, 22-section cast-iron radiator.

4. *Convectors.*—The convectors tested included one cast-iron convector, housed in a cabinet about 26 in. high by 57 in. long by 6⅞ in. wide, with outlet air grille in the top of the cabinet, shown in Figs. 2 and 3; and one non-ferrous convector, about 36 in. in length, shown in Figs. 4 and 5. This convector was housed in a cabinet about 26 in. high by 40 in. long by 5⅞ in. wide, with outlet air grille either in the front or in the top of the cabinet.

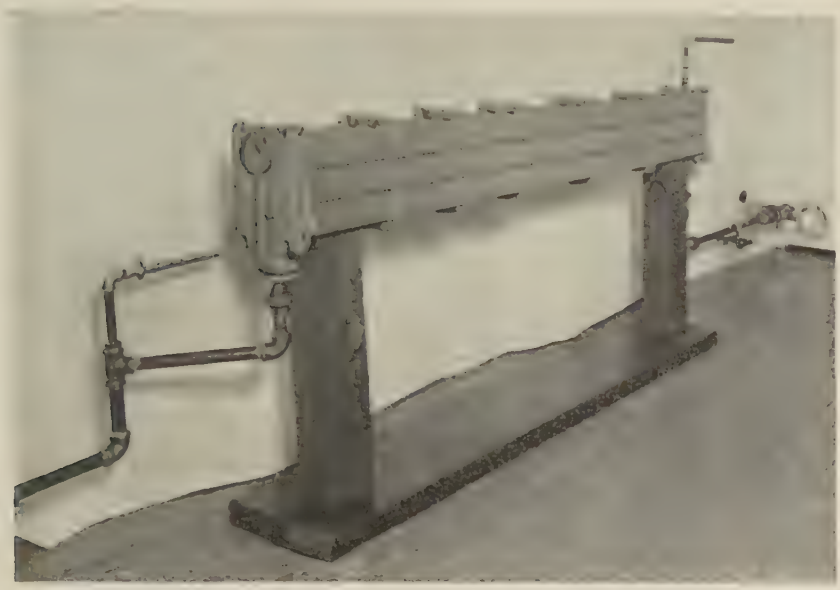

FIG. 2 HEATING ELEMENT OF CAST-IRON CONVECTOR WITH BOTTOM CONNECTIONS

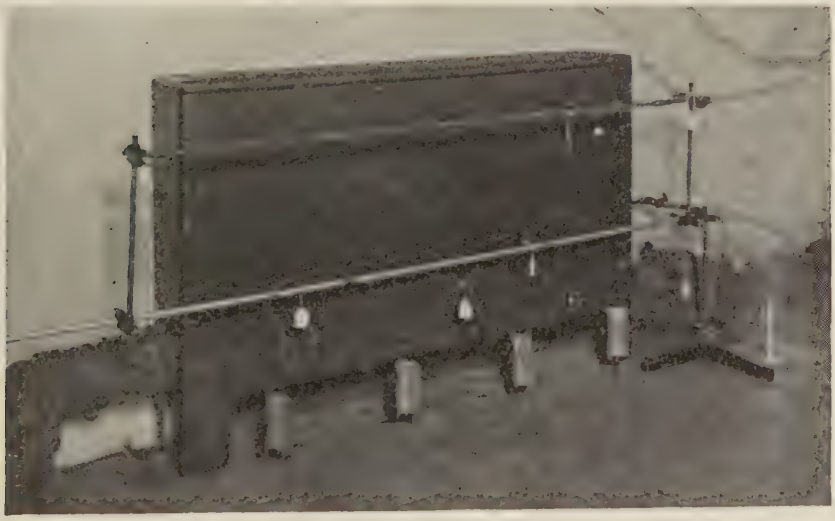

FIG. 3. ARRANGEMENT FOR TESTS ON HEAT EMISSION OF CONVECTOR

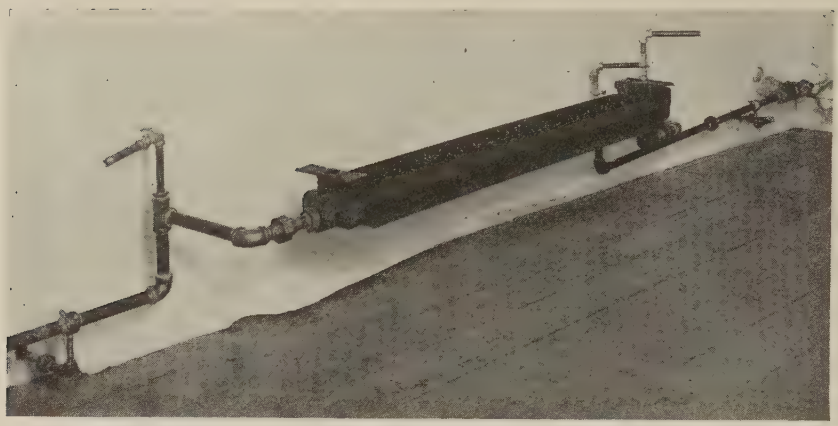

FIG. 4. HEATING ELEMENT OF NON-FERROUS CONVECTOR WITH END CONNECTIONS

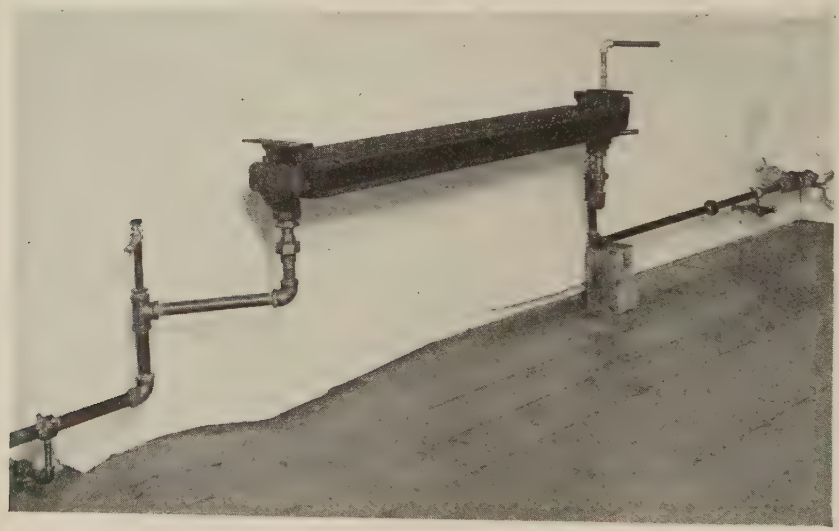

FIG. 5. HEATING ELEMENT OF NON-FERROUS CONVECTOR WITH BOTTOM CONNECTIONS

For the studies on friction heads the units tested were

(1) One cast-iron convector with end connections
(2) One cast-iron convector with bottom connections (Fig. 2)
(3) One non-ferrous convector with end connections (Fig. 4)
(4) One non-ferrous convector with bottom connections (Fig. 5).

III. Description of Apparatus and Methods of Testing

5. *Test Plant.*—For the purpose of the study, the test plant shown schematically in Fig. 6 was installed in the laboratory of the Department of Mechanical Engineering. The test booth is located on the third floor of the test plant. The radiator or convector to be tested was placed in the booth, which is enclosed on three sides, as shown in Fig. 7. In order to prevent extraneous air circulation, the open side, located opposite the wall adjacent to which the radiators or convectors

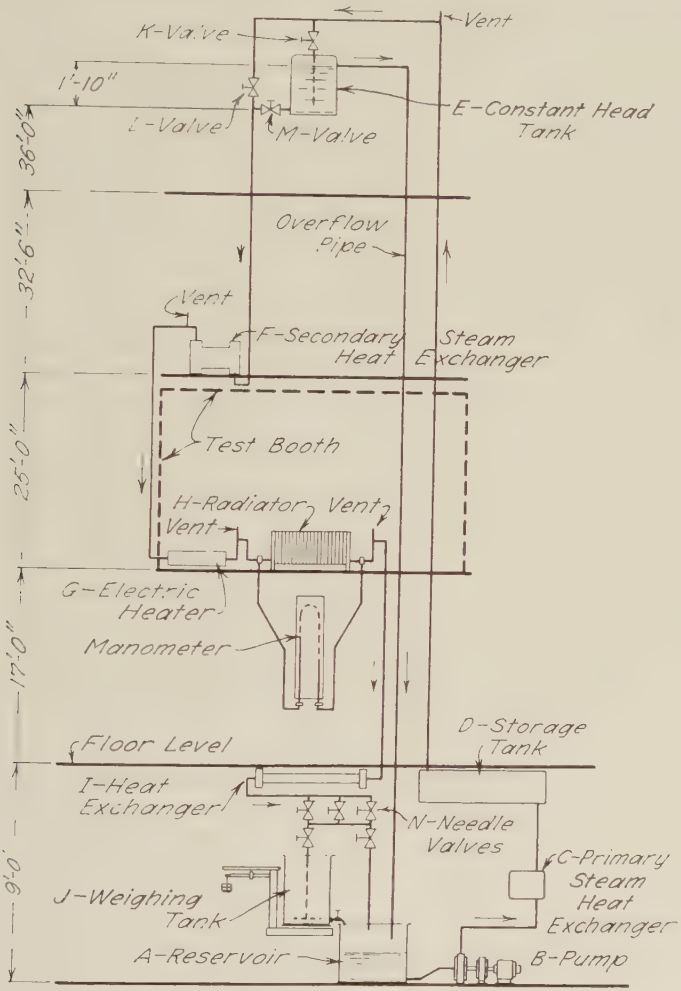

Fig. 6. Schematic Arrangement of Test Equipment

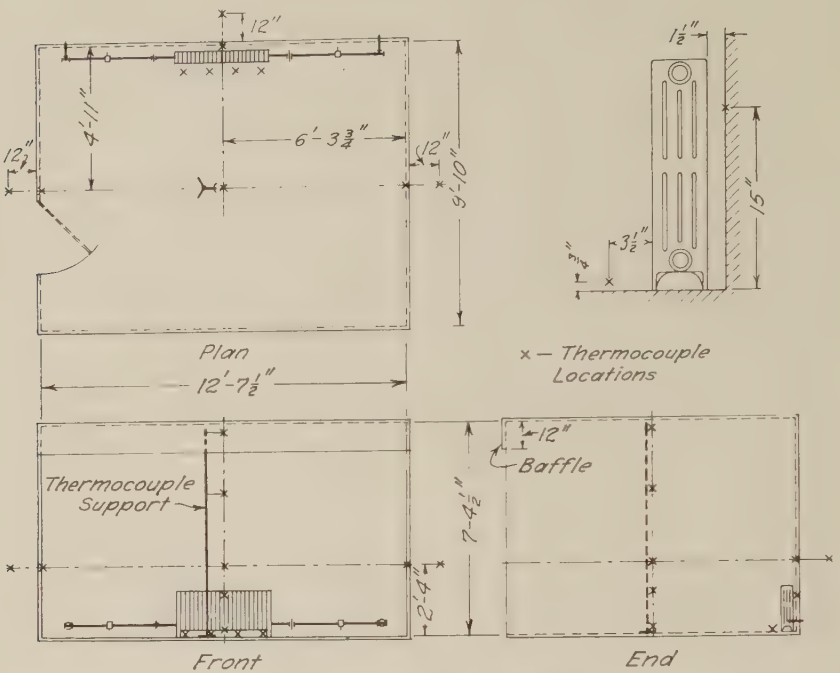

Fig. 7. Locations of Radiator and Thermocouples in Test Booth

were installed, is provided with a baffle extending downward 1.0 ft. from the ceiling. The booth is built of ½-in. wall board supported by a steel frame, and is not in contact with the floor, ceiling, or walls of the laboratory, so that it is completely surrounded by the laboratory air.

By opening the valves K and M, shown in Fig. 6, and by closing valve L, the test plant can be operated on gravity head furnished by water maintained at a constant level in the elevated constant-head tank E. By opening valve L, and closing valves K and M, the water from the pump B can be admitted directly into the line and circulated through the radiator or convector. Constant-head tank E is well insulated and provided with an overflow pipe serving to maintain a constant water level.

As the water flows from the pump B to the constant-head tank E it passes through a steam heat exchanger C, equipped with thermostatic control, and through a well-insulated storage or control tank D. A secondary heat exchanger F, and a thermostatically-controlled electric heater G, installed just before the radiator or convector, provide

the final stages of control, and permit the water to be heated to a temperature as high as 240 deg. F.

The pump and the overflow pipe have sufficient capacities so that the storage tank remains full under all possible rates of operation of the heating system.

6. *Method of Heating Water.*—As the water flows from the constant-head tank *E* to the radiator or convector *H*, it flows through the secondary steam heat exchanger *F*, and through the electric heater *G*. The heat exchanger *F* can be controlled either manually, or by maintaining the steam pressure at a definite value by means of a pressure-reducing valve. The electric heater has three heat rates, supplying, respectively, 1000, 2000, or 4000 watts at 115 volts. The voltage can be varied manually by means of a rheostat. The temperature of the water leaving the electric heater can also be controlled thermostatically within a range of temperatures from 100 deg. F. to 275 deg. F.

As the water flows from the radiator or convector *H* to the weighing tank *J*, it passes through the heat exchanger *I*, in which its temperature is reduced below the boiling point before it enters the weighing tank. Three needle valves of different sizes, shown at *N*, provide a means of controlling the rate of flow of the water through the radiator or convector.

7. *Method of Weighing Water.*—As the water enters the weighing tank it flows through a tee into two short horizontal branch pipes in order to reduce the effect of the impact on the weighing tank as much as possible. In practically all cases, the rate of flow of water through the radiator or convector was determined by finding the time required to collect 50 lb. of water in the weighing tank. To compensate for the buoyant effect of the vertical 1-in. pipe immersed in the water, it was necessary either to deduct from the scale reading of 50 lb. the weight of the water displaced by the pipe, which, for the tank *J* was 0.186 lb.; or to multiply the scale reading by 0.996. However, if the initial weight reading on the scale were taken before the level of the water in the tank had risen above the tee and its branch pipes, the scale reading would have to be corrected either by deducting the weight of the water displaced by the tee and its branch pipes which, in this case, was 0.243 lb.; or by multiplying the 50-lb. scale reading by 0.995 to compensate for the buoyant effect of the tee and its branch pipes, in addition to the buoyant effect of the additional vertical pipe.

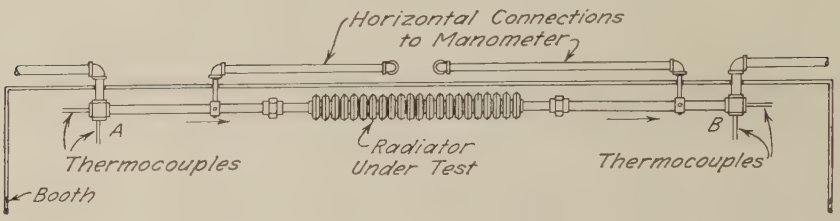

Fig. 8. Connection of Piezometer Rings With Manometer Tubes

8. *Method of Measuring Temperatures.*—All temperature readings were made by means of thermocouples and a potentiometer. The drop in temperature of the water as it passed through the radiator or convector was measured by means of four thermocouples, two of which were placed in the liquid stream at A and two at B, as shown in Fig. 8. The pipes leading to or from the radiator or convector were covered with two layers of 1-in. hair felt, as shown in Figs. 9 and 10, so that the heat emission from those pipes was so small that any resulting drop in temperature could not be measured.

Air temperatures in the central axis of the booth were measured at distances of 3 in., 18 in., 30 in., and 60 in. above the floor and 3 in. below the ceiling, as shown in Fig. 7. The temperature of the air in the booth was measured at four points also, ¾ in. above the floor, and 3½ in. in front of the radiator or convector. These thermocouples, as shown in Figs. 3, 9, 10, and 11, were shielded to eliminate the effect of radiation. Air temperatures one foot outside the booth were measured at points indicated in Fig. 7.

Temperatures of the surfaces of the floor and ceiling were meas-

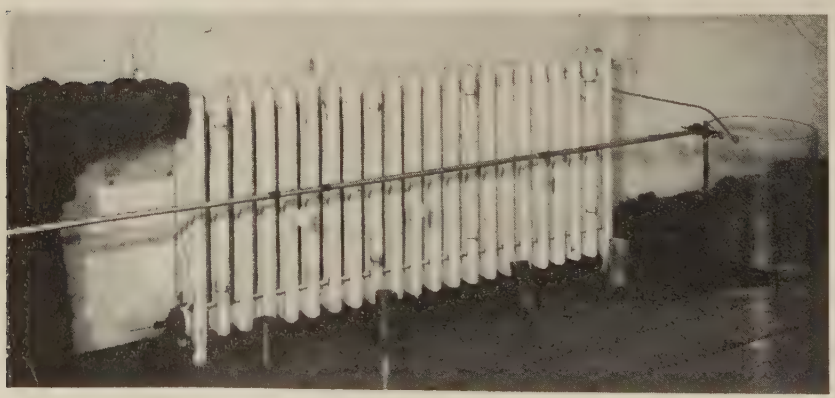

Fig. 9. 19-in., 4-tube, 22-section Radiator With Connection at Upper Tapping

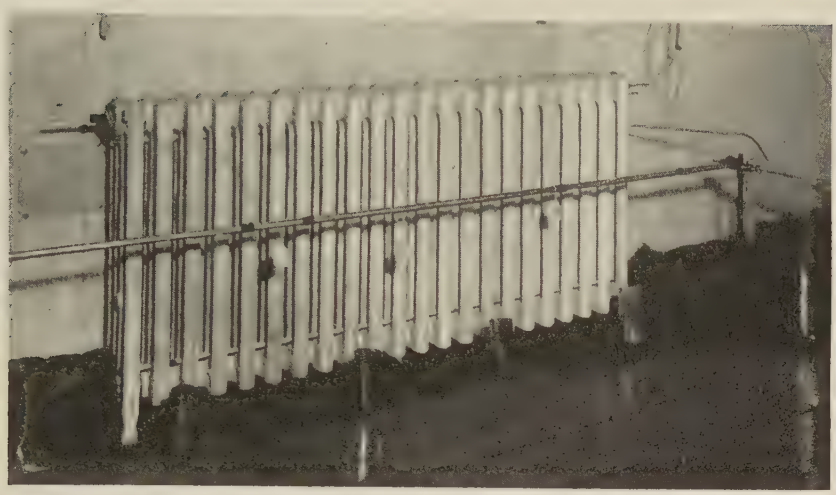

FIG. 10. 19-IN., 4-TUBE, 22-SECTION RADIATOR WITH CONNECTION AT LOWER TAPPING

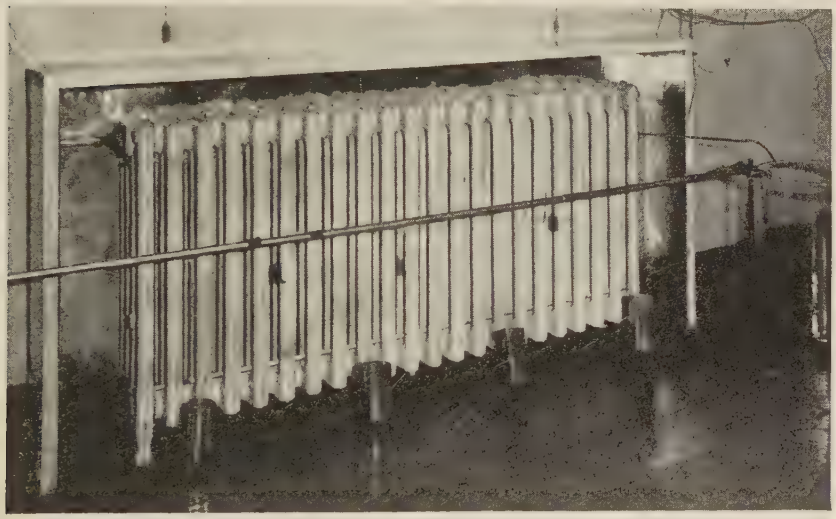

FIG. 11. 19-IN., 4-TUBE, 22-SECTION RADIATOR LOCATED IN RECESS SIMULATING WALL RECESS

ured in the central axis of the booth, and temperatures of the interior surfaces of the three walls were measured 30 in. above the floor and in the vertical axes of the respective walls. In addition, the temperature of the interior surface of the rear wall was obtained directly behind the radiator at a distance of 15 in. above the floor.

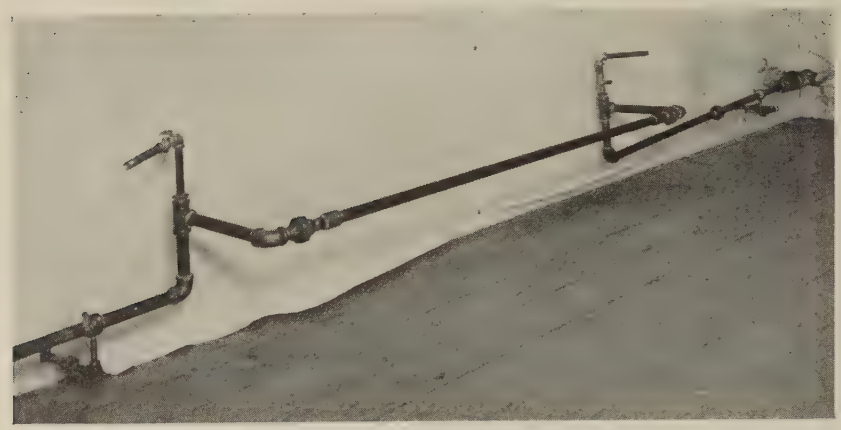

FIG. 12. PIPE REPLACING CONVECTOR IN FRICTION HEAD TESTS

9. *Method of Measuring Friction Heads.*—Friction heads were measured by means of manometers. During the earlier period of the study the manometers were connected to the piezometer rings on the pipe line leading to and from the radiator or convector, as shown in Figs. 2, 4, 5, and 12. Later the connection was changed by omitting the short vertical pipe leading downward from the piezometer rings. The error caused by these short vertical pipes could be about 0.025 in. of water when the drop in temperature through the radiator was 20 deg. F. The vertical manometer tubes were connected, as shown in Fig. 8, to the piezometer rings by means of horizontal 1-in. pipes over which air was blown continuously by means of a fan in order to cool the horizontal connections so that the temperature of the water in the two vertical manometer tubes would always be equal to that of the ambient air. Thus, the manometer reading could not be affected by a difference in temperature of the water in the two tubes.

10. *Method of Testing Radiators.*—The rate of flow of water through the radiator was determined by observing the time necessary to collect 50 lb. of water in the weighing tank. During this period

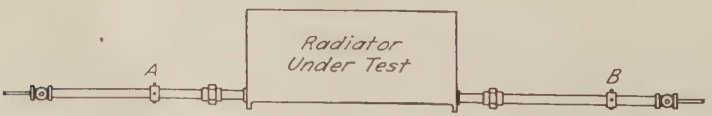

FIG. 13. ARRANGEMENT FOR TESTING FRICTION HEADS OF RADIATORS

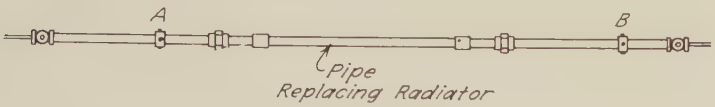

FIG. 14. ARRANGEMENT FOR TESTING FRICTION HEADS IN PIPES CONNECTED TO RADIATORS

observations were made of the temperatures of the water entering and leaving the radiator, and of the various air and wall-surface temperatures. The rate of heat transmission was then calculated from the rate of flow and the temperature drop of the water.

To determine the friction head in a radiator, an installation was made as indicated in Fig. 13, and the friction heads between the points A and B were measured as described in Section 9 for various rates of flow, when the temperature of the water was practically equal to that of the ambient air. The friction heads thus determined included both the friction heads in the radiator and the friction heads in the two sections of pipe between the piezometer rings and the radiator. The radiator was then replaced by a section of pipe, equal in length to that of the radiator, as shown in Fig. 14, and the test was repeated.

From the results of the second test the friction heads in the two sections of pipe joining the radiator to the piezometer rings were calculated and deducted from the friction heads determined in the first test; the remainders represented the friction heads in the radiator.

11. *Method of Testing Convectors.*—The rate of heat emission for convectors was determined by the same method as that used for radiators.

To determine friction heads of convectors, which could have inlet connections either at the end or at the bottom, the convectors were installed as shown in Figs. 2, 4, and 5. The short sections of pipe used in the installations were of such lengths that the same pieces of pipe could be used (but in different sequences) irrespective of whether the convector had an end inlet connection or a bottom inlet connection. Consequently, the test installation for a convector with end inlet included exactly the same length of pipe and the same number and kind of pipe fittings as the test installation for the convector with bottom inlet connections. Therefore any difference in the friction heads observed in two test installations was due entirely to the difference between the inlet constructions of the two convectors.

In order to determine what portion of the friction head of the test installation, shown in Fig. 4, for example, was the friction head of the

convector and what portion was the friction head of the connecting pipe and pipe fittings, the convector was removed and replaced by a pipe equal in length to that of the convector as shown in Fig. 12, and the test repeated.

The difference between the friction head of the installation shown in Fig. 4 and that of the installation shown in Fig. 12 was the difference between the friction head of the convector and that of the pipe used to replace the convector. The friction head of the pipe was determined as indicated in Fig. 14 and added to, or subtracted from, the difference between the friction heads of the installation shown in Fig. 4 and of that shown in Fig. 12 to determine the friction head of the convector.

IV. Heat Emission Rate of Radiators

12. *Course of Water Through Radiator.*—For this study, a 19-in., 4-tube, 22-section radiator with 1-in. push nipples and without cross bars was used.

It was found that when the water enters the radiator through the upper tapping (Fig. 9) the radiator may be considered as a two-pipe, reversed-return, forced-circulation heating system, as shown in Fig. 15.

The upper row of push nipples, with the intervening spaces, may be regarded as the flow main; the lower row of push nipples, with the intervening spaces, as the return main; and the 88 tubes, as 88 risers. For Test 269.3, the data of which are represented in Fig. 15, the flow main should carry water at a temperature of 217.1 deg. F., the return

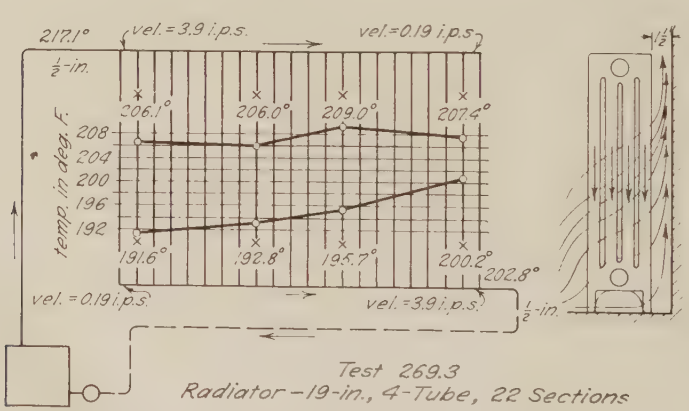

Fig. 15. Surface Temperatures for Radiator With Top Inlet and Bottom Outlet Connections

main water at a temperature of 202.8 deg. F., and the tubes water at temperatures decreasing gradually from 217.1 deg. F. to 202.8 deg. F. To determine whether this temperature distribution actually existed, eight thermocouples were attached to four outside tubes, as indicated by crosses in Fig. 15, in which the observed temperatures of the tubes are also presented. Near the upper ends of the tubes the temperatures were practically 207 deg. F., and near the lower ends the temperature increased gradually from 191.6 to 200.2 deg. F. These eight observed temperatures indicate

(1) that the 88 tubes function as return risers in the hypothetical system;

(2) that the circulation through the tubes increases as the distance from the inlet increases. This increase is the result of the variation in the velocity heads in the flow and return mains shown in Fig. 15;

(3) that considerably less than one-half of the water flows through the two outer rows of tubes. This decreased flow should be expected because the path from the flow main to the return main is longer through the outer tubes than through the inner tubes, as shown in Fig. 15.

It was found that when the water enters the radiator through the lower tapping (Fig. 10), the radiator may be considered a one-pipe heating system with forced circulation through the lower push nipples and gravity circulation in the tubes, provided the water cannot flow through the upper row of push nipples.

To prevent the flow of water through the upper row of push nipples, a brass rod 1.0 in. by 36 in. was inserted in this row before the radiator was tested. The average surface temperatures, for Tests 243-246, of the four outer tubes, near their upper ends, are shown in Fig. 16a. The resulting temperature gradient, shown by a heavy line in this figure, indicates that the radiator was functioning as a one-pipe heating system.

Without the brass rod in the radiator, the water flowing through the radiator from A to B, (Fig. 16b), may take either the direct and short path through the lower row of nipples, or the indirect and longer path from A upward through the first section to C; from there through the upper row of nipples to D; and from D downward through the last section to B.

The temperature gradient shown in Fig. 16b, determined from Tests 236-239, indicates that the larger quantity of water flows along the short path and that the radiator functions primarily as a one-pipe heating system, but that a portion of the water flows along the longer

path, from A to C, to D, and to B, and that, consequently, the radiator functions also, to a slight degree, as a two-pipe reversed-return heating system.

13. *Variation in Rate of Heat Emission With Temperature Difference—Water to Air.*—A study of the water temperatures indicated by the thermocouple readings shown in Figs. 15 and 16, and of the variable courses of the flow of water through the radiators, shows that it is practically impossible to determine the actual mean temperature of the water within the radiator and, therefore, it was decided to base all computations and analyses on a mean temperature equal to the average of the temperatures of the water entering and leaving the radiator. The latter will hereafter be referred to as the mean temperature of the water.

Similarly, it is impossible to determine the true mean temperature of the ambient air. This problem is further complicated by the fact that the heat emission from a radiator is partly by convection and partly by radiation, and that the heat emission by radiation varies with the mean radiant temperature of the space within which the radiator is located. The mean radiant temperature is difficult to determine, and is likely to differ by several degrees from the mean temperature of the ambient air. Therefore, in order to serve as a standard method for evaluating the rate of heat emission of a radiator, it was decided to substitute for the mean temperature of the ambient air the mean temperature of the air flowing toward the lower part of the radiator, as measured by thermocouples placed 3½ in. in front of the radiator and midway between the floor and the bottom of the

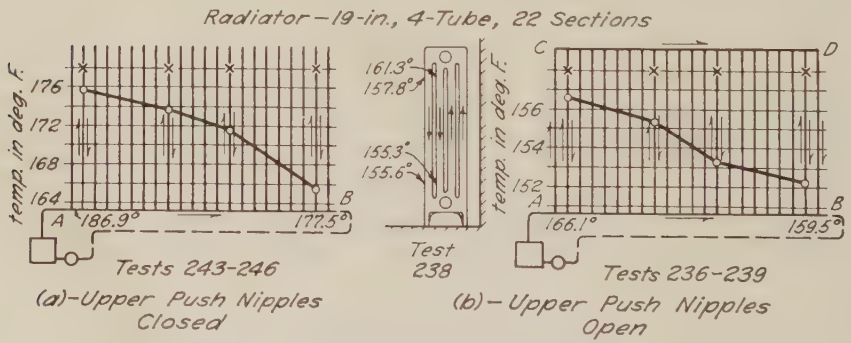

Fig. 16. Surface Temperatures for Radiator With Bottom Inlet and Outlet Connections

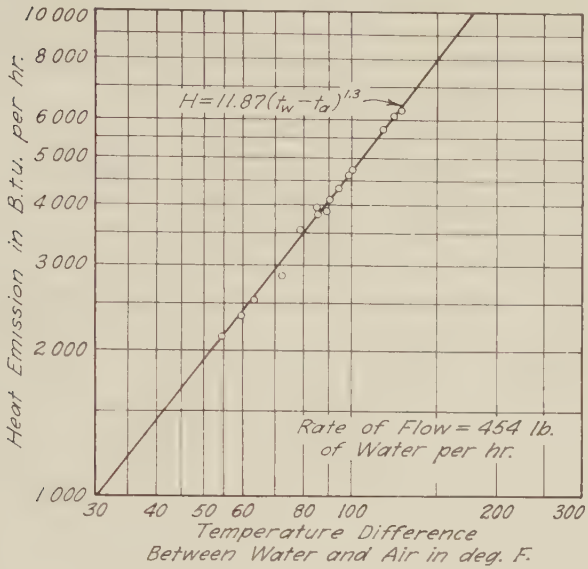

FIG. 17. RATE OF HEAT EMISSION OF 19-IN., 4-TUBE, 22-SECTION RADIATOR WITH TOP INLET AND BOTTOM OUTLET CONNECTIONS

radiator. These thermocouples were shielded from radiant energy emitted by the radiator, as shown in Figs. 3, 9, 10, and 11. All differences in temperature, water to air, have accordingly been based on the mean of the temperatures of the water entering and leaving the radiator, and on the mean temperature of the air $3\frac{1}{2}$ in. in front of the radiator.

The variation in heat emission with the difference in temperature, water to air, for a 19-in., 4-tube, 22-section radiator with water entering through the upper tapping is shown in Fig. 17, based on Test Nos. 256-263 and 268-269. The line drawn in Fig. 17 through the points determined experimentally, slopes at an angle, the tangent of which is 1.3. The slope of this line could be varied slightly and still represent the experimental data quite accurately. The slope of 1.3 was selected because earlier experiments indicated that the heat emission of radiators varies as the 1.3 power of the temperature difference, water to air, and because that relation has been quite generally accepted by the engineering profession.

The equation of the line shown in Fig. 17 is

$$Y = 11.87\,X^{1.3} \qquad (1)$$

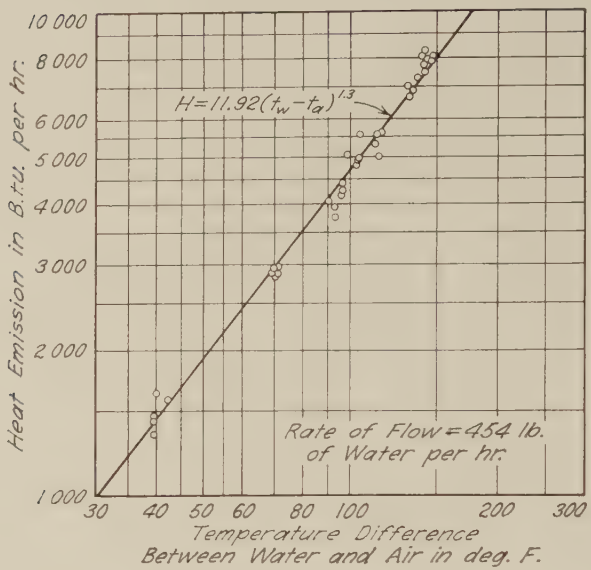

Fig. 18. Rate of Heat Emission of 19-in., 4-tube, 22-section Radiator With Bottom Inlet and Outlet Connections

Hence, according to the experimental data shown in Fig. 17, the rate of heat emission of the radiator tested (Fig. 9) was

$$H = 11.87\ (t_w - t_a)^{1.3} \tag{2}$$

in which

H = heat emission, B.t.u. per hr.

t_w = mean water temperature, deg. F.

t_a = mean air temperature, deg. F.

When the radiator was tested with water entering through the lower tapping (Fig. 10) the variation in the rate of heat emission with the temperature difference was found to be as shown in Fig. 18, based on Tests No. 169-204. The line showing average results was again drawn at an angle having a tangent of 1.3. The equation of this line is

$$Y = 11.92\ X^{1.3} \tag{3}$$

Hence, for this case, the rate of heat emission was

$$H = 11.92\ (t_w - t_a)^{1.3} \tag{4}$$

The difference between the coefficients in the two equations is only one-half of one per cent. This deviation is easily within the limits of experimental error, and it may be concluded that the rate of heat emission is the same for water entering through the upper tapping as for water entering through the lower tapping. For this particular radiator the equation may therefore be written:

$$H = 11.90\ (t_w - t_a)^{1.3} \tag{5}$$

14. *Variation in Rate of Heat Emission With Variation in General Form or Design of Radiators.*—For this study, two radiators were selected which differed only in that one had 1¼ in., instead of 1 in., push nipples. This radiator also had slightly heavier tubes than the other, so that the distances between tubes were slightly smaller, and had one row of cross bars. Otherwise the two radiators were alike.

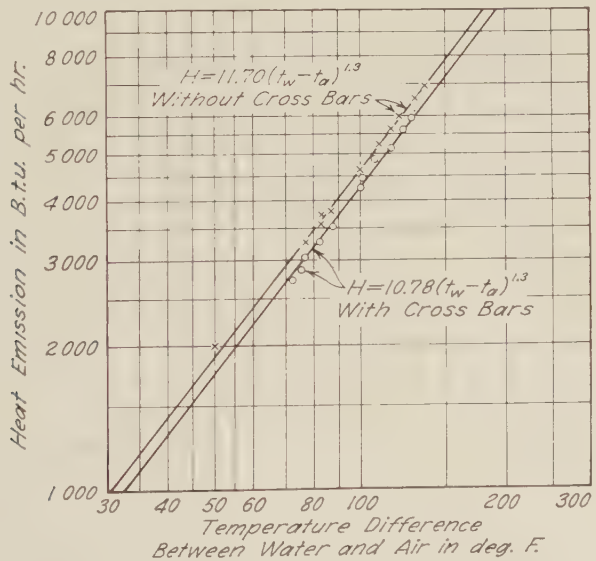

FIG. 19. COMPARISON OF RATES OF HEAT EMISSION OF TWO 19-IN., 4-TUBE, 22-SECTION RADIATORS DIFFERING SLIGHTLY IN GENERAL FORM OR DESIGN

The two radiators were tested under identical conditions, and the test results are shown in Fig. 19 in which the crosses represent the values found for the radiator without cross bars and the circles represent the values found for the radiator with cross bars.

The equations for heat emission, deduced from the lines representing average values are, respectively,

$$H = 11.70\ (t_w - t_a)^{1.3} \tag{6}$$

and

$$H = 10.78\ (t_w - t_a)^{1.3} \tag{7}$$

Hence the heat emission of the radiator with cross bars was approximately 8 per cent lower than that of the radiator without cross bars.

15. *Influence of Distance Between Tubes and of Cross Bars on Rate of Heat Emission.*—It was thought that a change in the distance between tubes might affect the rate of heat removal by convection currents and thereby affect the rate of heat emission, and that the introduction of cross bars between tubes might affect the course of the water through the radiator and thereby reduce the mean temperature of the water in the outer row of tubes.

To study the influence of the distance between tubes on heat emission, ammonium chloride fumes were liberated from shallow dishes placed on the floor near the bottom of the radiator. As shown in Fig. 15 it was observed that the space between the radiator and the wall acted as a chimney. The main stream of air entering the radiator near the bottom flowed in an inclined upward direction through the spaces between the radiator sections and then passed upward between the radiator and the wall. Thus air from near the floor did not pass over the upper half of the front row of tubes. Previous studies,* however, indicated that some air from the room was drawn over the upper half of the front tubes, passed up between the sections, and was delivered out of the top of the radiator. The evidence seemed to indicate that the presence of cross bars did not interfere with the air flow sufficiently to influence the heat emission appreciably.

The distance between the radiator and the wall was 1½ in. A variation in this distance may have an appreciable effect on the heat emission of the radiator, and there may be an optimum distance between tubes for the removal of heat by convection currents. No tests

*"Investigation of Various Factors Affecting the Heating of Rooms With Direct Steam Radiators." Univ. of Ill. Exp. Sta. Bul. No. 223, pp. 86 and 87.

were made to determine the optimum distance from the radiator to the wall for a maximum rate of heat emission, or the optimum distance between tubes.

To study the influence of cross bars on heat emission, the system of piping shown in Fig. 20 was prepared and tested. It was assumed that with water flowing from the point A to the point B by either of two paths, ACB or ADB, the larger portion of the water would take the shorter path, ACB; and, if the two tubes were of equal size, the velocity head at C would be greater than that at D. Therefore, if a connection existed between the points D and C, some of the water would flow from D to C and, consequently, a smaller quantity would flow from D to B, resulting in a lower velocity and a lower average temperature from D to B. The average surface temperature of the tube ADB should, therefore, be lower if a cross bar existed than if no cross bar existed. Hence the heat emission of a radiator, with cross bars between its tubes, should be lower than that of the same radiator, without cross bars.

It was considered that the pipes ACB might represent two inner

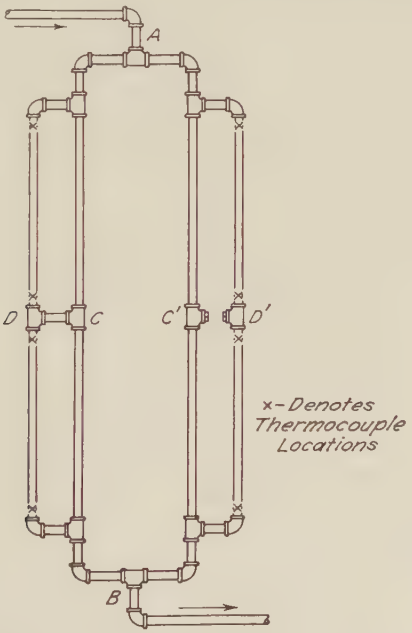

FIG. 20. ARRANGEMENT OF PIPING FOR TESTS ON EFFECT OF CROSS BARS ON RATE OF HEAT EMISSION

TABLE 1
TEMPERATURE IN PIPING ARRANGEMENT MODEL

Test No.	Average Temperature of Outside Tubes, Deg. F.	
	With Cross Bars	Without Cross Bars
1	145.4	147.0
2	200.2	201.7
3	200.2	202.6
4	207.5	210.3
5	215.4	218.3
6	217.6	220.5
Average	197.7	200.1

tubes of a 4-tube radiator section and the pipes ADB might represent two outer tubes of the section, one set of tubes having a cross bar and the other having none. The apparatus was constructed so that the friction in the two lines should be as nearly equal as possible, in order that equal volumes of water could flow through the two parts.

It was realized that the two tees, which were necessary to install the cross bars, would interfere with the uniform flow of water assumed in Fig. 20, but it was thought that comparative results might be secured in spite of this interference.

Eight thermocouples were attached to the apparatus at the points marked by crosses. Six tests were made and the average temperatures obtained on the outer pipes or tubes are given in Table 1.

The difference between the two was only 1.2 per cent, but the results do indicate a difference definitely in favor of the radiator without the cross bars.

16. *Ratio of Rate of Heat Emission of an End Section to That of an Intermediate Section.*—For this study, one of the 22-section radiators previously tested was changed to a 12-section radiator by removing ten of the twenty intermediate sections. By testing the 12-section radiator in comparison with the 22-section radiator, it was found that the rate of heat emission of an end section is 1.275 times that of an intermediate section. Hence, for design purposes, the two end sections may be considered equal to 2.5 sections.

17. *Influence of Rate of Flow of Water Through Radiator on Rate of Heat Emission.*—When the 19-in., 4-tube, 22-section radiator was tested and found to emit heat at the rate shown by Equation (2),

water was flowing through the radiator at the rate of 454 lb. per hr., resulting in a velocity of 11.6 in. per sec. in the radiator connections. When the rate of flow was increased to 1816 lb. per hr., the rate of heat emission was reduced almost 6.0 per cent. No doubt the reason for this reduction in the rate of heat emission was that, with a higher velocity, a larger proportion of the total flow of water passed through the two inner tubes. Consequently, the temperature of the outer tubes was not increased in proportion to the increase in the mean temperature of the water in the radiator; that is, in proportion to the average of the temperatures of the entering and leaving water, this average having been arbitrarily defined as the mean temperature of the water in the radiator.

18. *Ratio of Rate of Heat Emission of a Radiator Located in a Wall Recess to That of a Free-standing Radiator.*—For this study, a recess similar to those used in the I=B=R Research Home at the University of Illinois was constructed. The recessed radiator is shown in Fig. 11. The clearances between the ends of the radiator and the sides of the recess were each 4 in., that between the top of the radiator and the top of the recess was 2¼ in., and that between the back of the radiator and the back surface of the recess was ¼ in. With the radiator thus located, the rate of heat emission, H, in B.t.u. per hr. was found to be

$$H = 1.90\ (t_w - t_a)^{1.65} \tag{8}$$

The ratios of the rate of heat emission of the 19-in., 4-tube, 22-section radiator located in a recess to that of the free-standing radiator, for various temperature differences, water to air, were calculated by means of Equations (8) and (2), and are shown in Table 2.

TABLE 2
RATIOS OF HEAT EMISSION RATES FOR RECESSED RADIATORS

Temperature Difference $t_w - t_a$ deg. F.	Ratios of Heat Emission Rates, per cent	
	Free-standing Radiator	Radiator in Recess
70	100	71
85	100	76
100	100	80
115	100	85
130	100	87

19. *Variation of Rate of Heat Emission With Variations in Height of Radiator and Number of Tubes per Section.*—For this study, the five radiators shown in Fig. 1 were tested, and the following equations for the rate of heat emission, H, in B.t.u. per hr. were derived:

19-in., 4-tube, 16-section radiator

$$H = 8.9\ (t_w - t_a)^{1.3} \tag{9}$$

22-in., 4-tube, 16-section radiator

$$H = 10.59\ (t_w - t_a)^{1.3} \tag{10}$$

25-in., 4-tube, 16-section radiator

$$H = 11.79\ (t_w - t_a)^{1.3} \tag{11}$$

25-in., 3-tube, 16-section radiator

$$H = 9.18\ (t_w - t_a)^{1.3} \tag{12}$$

25-in., 5-tube, 16-section radiator

$$H = 13.55\ (t_w - t_a)^{1.3} \tag{13}$$

The rates of heat emission shown in Equations (9)-(13) were studied from various angles and it was found that, for radiators of similar design, there is a fairly definite relation between the heat emission of a radiator and the volume of the rectangular envelope enclosing the radiator. Hence there is also a definite relation between the heat emission and the product obtained by multiplying the height of the radiator by the number of tubes per section. It was shown in Section 16 that the heat emission of the end section of a radiator is approximately equal to 1.25 times that of an intermediate section. Making use of these relations the following formula was developed:

$$H = 0.014\ (nh)^{0.85}\ (t_w - t_a)^{1.3}\ (S + \tfrac{1}{2}) \tag{14}$$

in which

H = heat emission in B.t.u. per hr.

h = height of radiator in inches

n = number of tubes per section

S = number of sections per radiator.

To check the formula, the rates of heat emission of the five radiators shown in Fig. 1 were calculated and are given in Table 3 in comparison with the rates determined experimentally.

TABLE 3
RELATIVE HEAT EMISSION RATES OF SMALL-TUBE, 16-SECTION RADIATORS

Size of Radiator	Heat Emission Rate, B.t.u. per hr. per deg. F. temperature difference		Difference per cent
	Experimental	Calculated	
25-in., 5-tube	13.55	14.02	+3.4
25-in., 4-tube	11.79	11.55	−2.1
25-in., 3-tube	9.18	9.07	−1.2
22-in., 4-tube	10.59	10.39	−1.9
19-in., 4-tube	8.9	8.74	−1.8

The calculated rates differ from the rates determined experimentally by less than 4 per cent. It was shown in Section 14 that two radiators of equal height and with the same number of tubes, but differing from each other in minor structural details, may have heat emission rates differing by as much as 7 or 8 per cent. Hence, in the absence of more definite data from the radiator manufacturers, it seems feasible to use Equation (14) to calculate the approximate rates of heat emission of small-tube cast-iron radiators similar to those used in the tests.

20. *Comparison of Rates of Heat Emission of Radiator Operating With Hot Water and With Steam.*—For this study, the 19-in., 4-tube, 22-section radiator shown in Figs. 9 and 10 was selected.

Functioning as a hot water radiator, with water flowing through the radiator at a rate of 454 lb. per hr., the rate of heat emission had been determined to be $11.90\ (t_w - t_a)^{1.3}$ B.t.u. per hr. (Figs. 17 and 18 and Equation (5)).

To determine the rate of heat emission with the radiator functioning as a steam radiator, eight tests were made. The results are shown in Fig. 21. The line AB was drawn at an angle such that the tangent with the horizontal was 1.3. The line AB evidently represents the eight experimental values with sufficient accuracy, and is expressed by the equation

$$H = 12.48\ (t_w - t_a)^{1.3} \qquad (15)$$

The equation representing the thirty-eight experimental values determined for the same radiator when it was tested as a hot water radiator is

$$H = 11.90\ (t_w - t_a)^{1.3} \qquad (16)$$

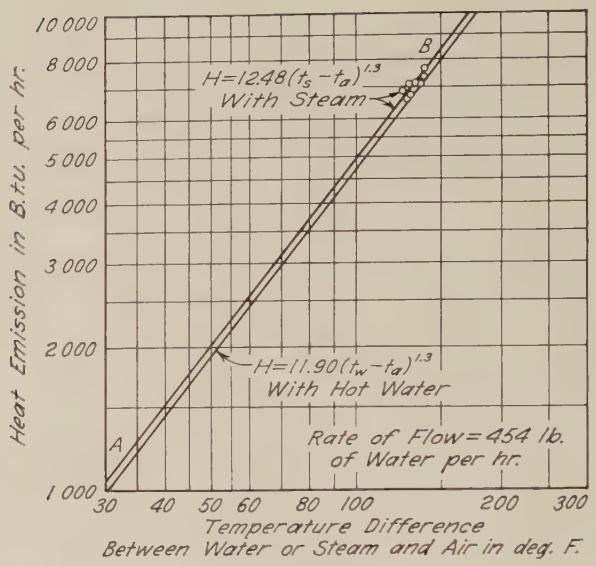

FIG. 21. COMPARISON OF RATES OF HEAT EMISSION OF A 19-IN., 4-TUBE, 22-SECTION RADIATOR OPERATING WITH HOT WATER AND WITH STEAM

The increase in the quantity of heat emitted by this radiator functioning as a steam radiator over that obtained with it functioning as a hot water radiator is $\frac{12.48 - 11.90}{11.90}$, or 5.0 per cent.

The difference of 5.0 per cent between steam radiation and hot water radiation found in the heat emission of the 19-in., 4-tube, 22-section radiator, with water flowing through it at the rate of 454 lb. per hr., does not strictly apply to all small-tube cast-iron radiators, because the tests described in this Section are in effect a comparison of the methods of designing steam radiation and hot water radiation. The heat emitted by a radiator varies with the temperature of the exterior surface. In the design of steam radiation, it is assumed that the steam temperature represents the mean temperature of the heating medium, and that the surface temperature is practically equal to the steam temperature. In this case these assumptions are reasonably accurate. In the design of hot water radiation, it is assumed that the temperature of the exterior surface is practically the same as the average temperature of the water within the radiator, and that the latter is equal to the mean of the temperatures of the water entering and leaving the radiator. These assumptions may be quite inaccurate,

as shown by the several radiator surface temperatures presented in Figs. 15 and 16.

The difference between the exterior surface temperature and the mean of the temperatures of the water entering and leaving the radiator varies with the velocity of the water and with the width of the radiator. If the water enters the radiator through the lower tapping and with a high velocity, a considerable portion will pass out of the radiator without having circulated through the tubes, and the average temperature of the water within the radiator will be lower than the mean of the temperatures of the water entering and leaving. The difference between these two temperatures is greater for wide radiators than for narrow radiators. If the water enters the radiator through the upper tapping and with a low velocity, as is the case in gravity-circulation systems, practically all of the water which flows through the radiator will circulate through the tubes, and the average temperature of the water within the radiator will be very nearly equal to the mean of the temperatures of the water entering and leaving; this is especially true in the case of very narrow radiators.

On the whole it seems safe to conclude (a) that the rate of heat emission of a 19-in., 4-tube, 22-section radiator with a rate of flow of 454 lb. of water per hr. is about 5 per cent higher with the radiator functioning as a steam radiator than it is with it functioning as a hot-water radiator; (b) that the difference is greater than 5 per cent for rates of flow larger than 454 lb. per hr. and smaller than 5 per cent for rates smaller than 454 lb. per hr.; (c) that, other conditions being the same, the difference between the rates of heat emission of a radiator functioning as a steam radiator or as a hot-water radiator is greater for wide than it is for narrow radiators; and (d) that, for very low rates of flow, especially when the water enters the radiator through the upper tapping, the difference, if any, between the rates of heat emission of a radiator functioning as a steam radiator or as a hot-water radiator is very small.

V. Heat Emission Rates of Convectors

21. *Variation in Rate of Heat Emission of Non-ferrous Convector With Temperature Difference—Water to Air.*—The convectors were installed as indicated in Fig. 3, and the tests were conducted in the same manner as those for the 4-tube radiator discussed in Section 10. It was found that, with water flowing through the convector at the

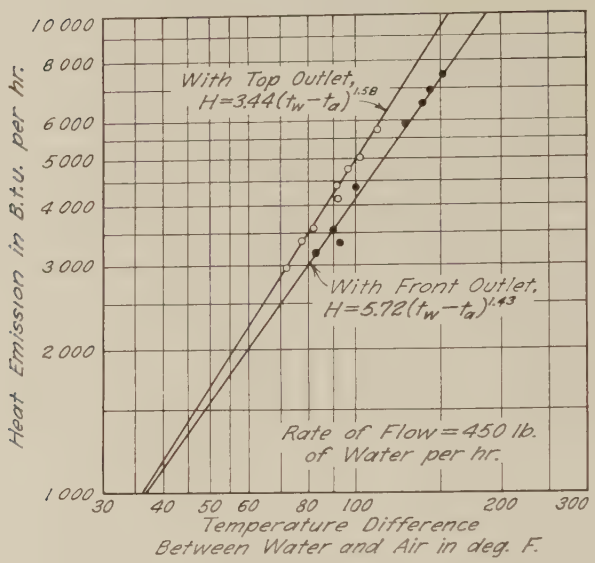

FIG. 22. RATE OF HEAT EMISSION OF NON-FERROUS CONVECTORS WITH FRONT AND TOP AIR OUTLETS

rate of 450 lb. per hr., the rate of heat emission of the non-ferrous convector, with air outlet grille in front, was

$$H = 5.72\ (t_w - t_a)^{1.43} \tag{17}$$

and, with air outlet grille in the top, was

$$H = 3.44\ (t_w - t_a)^{1.58} \tag{18}$$

The results are shown in Fig. 22.

The heat emission of the convector for the two cases calculated from Equations (17) and (18) are given in Table 4.

22. *Influence of Velocity of Water Flowing Through Non-ferrous Convector on Rate of Heat Emission.*—For this study, a cabinet with a horizontal air grille in the top was used in connection with the convector, and the results are shown as lines AB, CD, and EF in Fig. 23. It was found that when the rate of flow of water was increased from 450 to 888 lb. per hr., or practically doubled, the rate of heat emission was increased about 4 per cent. When the rate of flow was increased from 450 to 1708 lb. per hr., or practically quadrupled, the rate of heat emission was increased about 8 per cent.

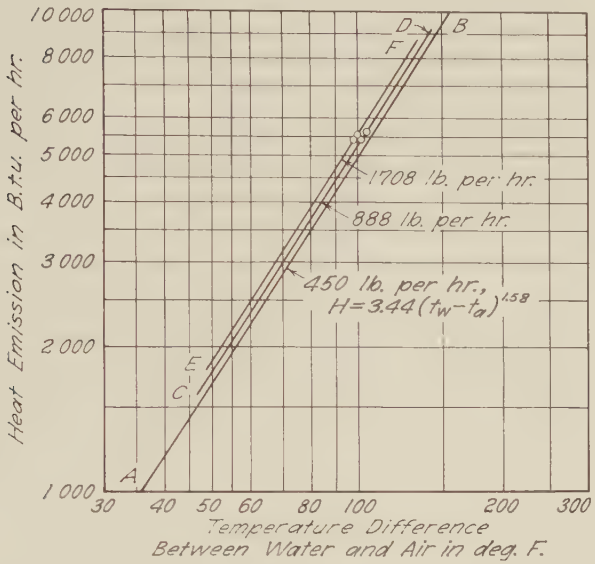

FIG. 23. COMPARISON OF RATES OF HEAT EMISSION OF NON-FERROUS CONVECTOR WITH TOP AIR OUTLET AT DIFFERENT RATES OF FLOW OF WATER

The increase in heat emission, with an increase in the velocity of the water through the convector undoubtedly resulted from the increased scrubbing effect of the water at higher velocities on the surface films of the three small pipes in the convector.

23. *Comparison of Rates of Heat Emission of Non-ferrous Convector Operating With Hot Water and With Steam.*—For this study, the cabinet with an outlet air grille in the top was used, and the results are shown in Fig. 24. The line represents the heat emission rate

TABLE 4
HEAT EMISSION RATE OF CONVECTORS WITH TOP AIR OUTLET AND WITH FRONT AIR OUTLET

Temperature Difference $t_w - t_a$ deg. F.	Heat Emission Rate, B.t.u. per hr.	
	Top Outlet	Front Outlet
70	2830	2490
85	3850	3280
100	4970	4140
115	6200	5060
130	7530	6030
145	8940	7050

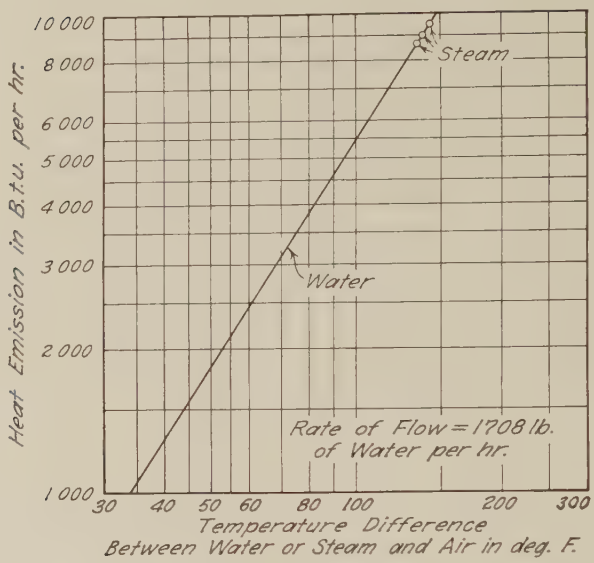

Fig. 24. Comparison of Rates of Heat Emission of Non-ferrous Convector Operating With Hot Water and With Steam

of the convector when functioning as a hot-water convector with water flowing through it at the rate of 1708 lb. per hr., or 3.4 gal. per min. The points represent the tests with the convector functioning as a steam convector. It may be observed that when operating with steam the rate of heat emission was practically the same as that obtained with a high rate of water flow through the convector.

24. *Variation in Rate of Heat Emission of Cast-iron Convector With Temperature Difference—Water to Air.*—For the heat emission tests on the cast-iron convector a cabinet having an outlet air grille in the top was used. The tests were conducted in the same manner as those for the cast-iron radiators discussed in Section 10. The results with water flowing through the convector at a rate of 437 lb. per hr. are shown in Fig. 25. The equation for the rate of heat emission is

$$H = 12.40\ (t_w - t_a)^{1.33} \qquad (19)$$

The heat emission rates, calculated from Equation (19) for various temperature differences, are given in Table 5.

25. *Influence of Velocity of Water Flowing Through Cast-iron Convector on Rate of Heat Emission.*—For this study, two additional

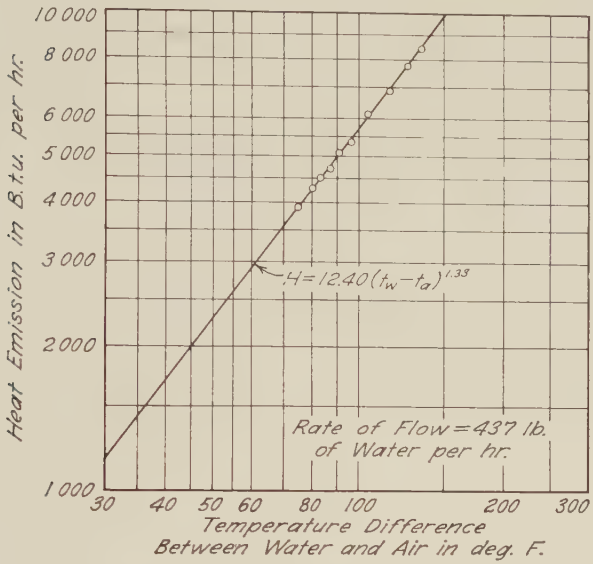

FIG. 25. RATE OF HEAT EMISSION OF CAST-IRON CONVECTOR WITH TOP AIR OUTLET

series of tests were made. In the first series, the rate of flow was increased from 437 to 905 lb. per hr., and in the second series from 437 to 1814 lb. per hr. The results are shown in Figs. 26 and 27, respectively, in which the base curves, shown as full lines, are for a water rate of 437 lb. per hr., and the points, shown as circles, are for the rates of 905 and 1814 lb. per hr. The curves indicate that increasing the rate of flow of the water through the convector, even to the extent of quadrupling it, had no appreciable effect on the rate of heat emission.

The experimental data shown in Fig. 27 are not as consistent as those shown in Figs. 25 and 26, probably because the temperature

TABLE 5
RATE OF HEAT EMISSION OF A CAST-IRON CONVECTOR

Temperature Difference $t_w - t_a$ deg. F.	Rate of Heat Emission, B.t.u. per hr.
70	3530
85	4570
100	5670
115	6830
130	8030
145	9290

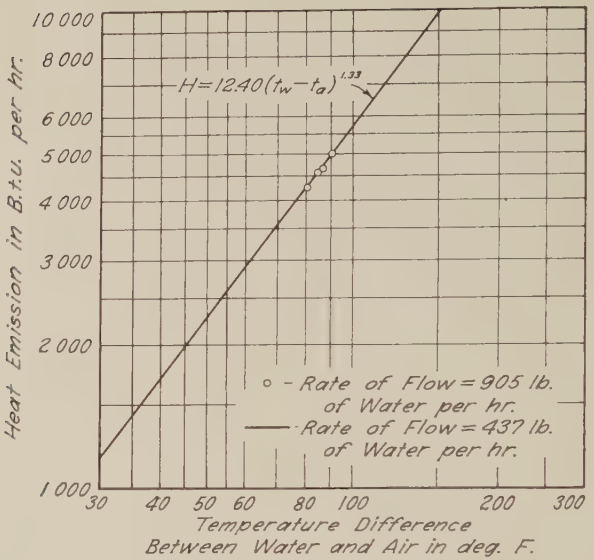
10 000
8 000
6 000
5 000
4 000
3 000
2 000
1 000
Heat Emission in B.t.u. per hr.
H=12.40(tw−ta)1.33
o - Rate of Flow = 905 lb. of Water per hr.
— Rate of Flow = 437 lb. of Water per hr.
30 40 50 60 80 100 200 300
Temperature Difference Between Water and Air in deg. F.

Fig. 26. Rate of Heat Emission of Cast-iron Convector With Rate of Flow of Water Doubled

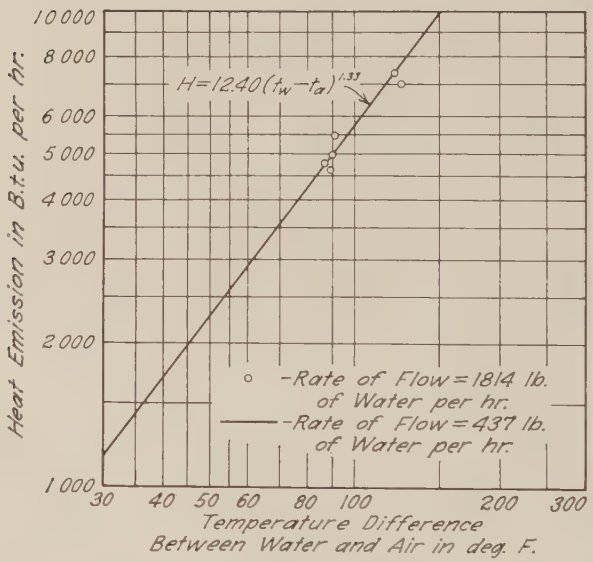
10 000
8 000
6 000
5 000
4 000
3 000
2 000
1 000
Heat Emission in B.t.u. per hr.
H=12.40(tw−ta)1.33
o -Rate of Flow = 1814 lb. of Water per hr.
— -Rate of Flow = 437 lb. of Water per hr.
30 40 50 60 80 100 200 300
Temperature Difference Between Water and Air in deg. F.

Fig. 27. Rate of Heat Emission of Cast-iron Convector With Rate of Flow of Water Quadrupled

drop through the convector could not be measured with sufficient accuracy. With water flowing through the convector at the rates of 437, 905, and 1814 lb. per hr., the respective average temperature drops through the convector were 12.8, 4.1, and 3.1 deg. F. An error of one-tenth of one degree in the temperature drop at the high velocity would result in an error of 3 per cent in the final result, whereas the same error at the low velocity would result in an ultimate error of only approximately eight-tenths of one per cent.

Within the range of velocities obtaining in practice the heat emission of a cast-iron convector is apparently independent of the velocity of the water flowing through the convector.

26. *Comparison of Rates of Heat Emission of Cast-iron Convector Operating With Hot Water and With Steam.*—Three tests were made and it was found that, with the cast-iron convector functioning as a steam convector at temperature differences, steam to air, of 140.9, 142.3, and 136.6 deg. F. the respective rates of heat emission were 8864, 9484, and 9021 B.t.u. per hr. The average of the three test results is 9123 B.t.u. per hr. for a temperature difference of 140 deg. F.

With the same convector, functioning as a hot water convector, at a temperature difference of 140 deg. F., water to air, the heat emission, as given by Equation (19), was 8867 B.t.u. per hr.

Hence, in this case, the rate of heat emission of the cast-iron convector, functioning as a steam convector, was 2.8 per cent higher than that for the same convector functioning as a hot water convector.

VI. Friction Heads of Radiators

27. *Friction Head of 19-in., 4-tube, 22-section Radiator With 1¼-in. Connections; Water Temperature 75 deg. F.*—At a water temperature of 75 deg. F., the friction heads of the arrangement shown in Fig. 13 were determined experimentally for various water velocities, using the method described in Section 10, and the results are presented as line AB in Fig. 28. The equation of this line is

$$H_f = 0.000562P^{1.88} \tag{20}$$

in which

H_f = friction head in milinches* of water for the radiator plus 4.708 ft. of 1¼-in. pipe.

P = rate of flow of water through the radiator in lb. per hr.

*One milinch = 0.001 inch.

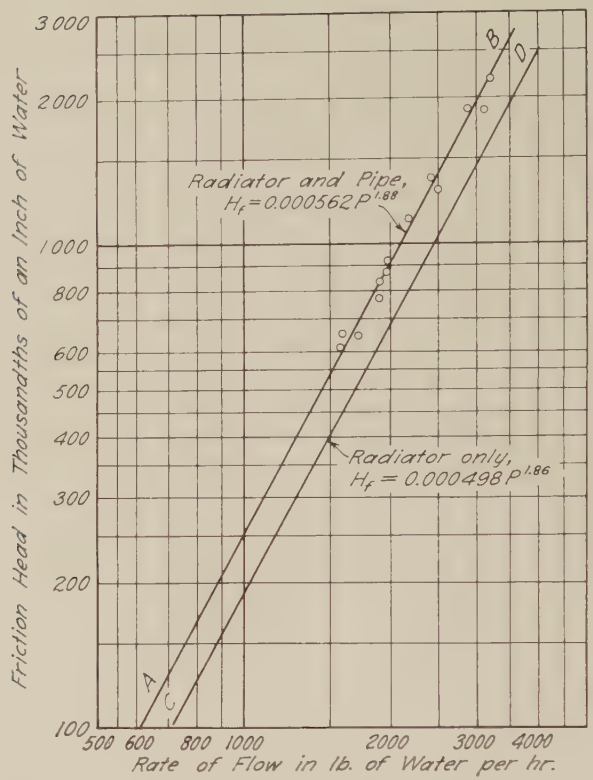

Fig. 28. Friction Head of 19-in., 4-tube, 22-section Radiator With 1¼-in. Connections

In order to find the friction head for 4.708 ft. of pipe, an installation similar to that shown in Fig. 14 was used. From the test results, the friction heads of 4.708 ft. of 1¼-in. pipe were calculated and deducted from those given by the line AB in Fig. 28. The remainders were the friction heads of the radiator. These heads are represented by the line CD in Fig. 28, the equation of which is

$$H_f = 0.000498P^{1.86} \tag{21}$$

The friction heads for the radiator with 1¼-in. connections, calculated from Equation (21), are shown as curve 1 in Fig. 29. As indicated by this curve, the friction head of the radiator was about 490 mi. with the water flowing through the radiator at a rate of 1500 lb. per hr.; that is, with a water velocity of about 0.65 ft. per sec. in the 1¼-in. pipe connection.

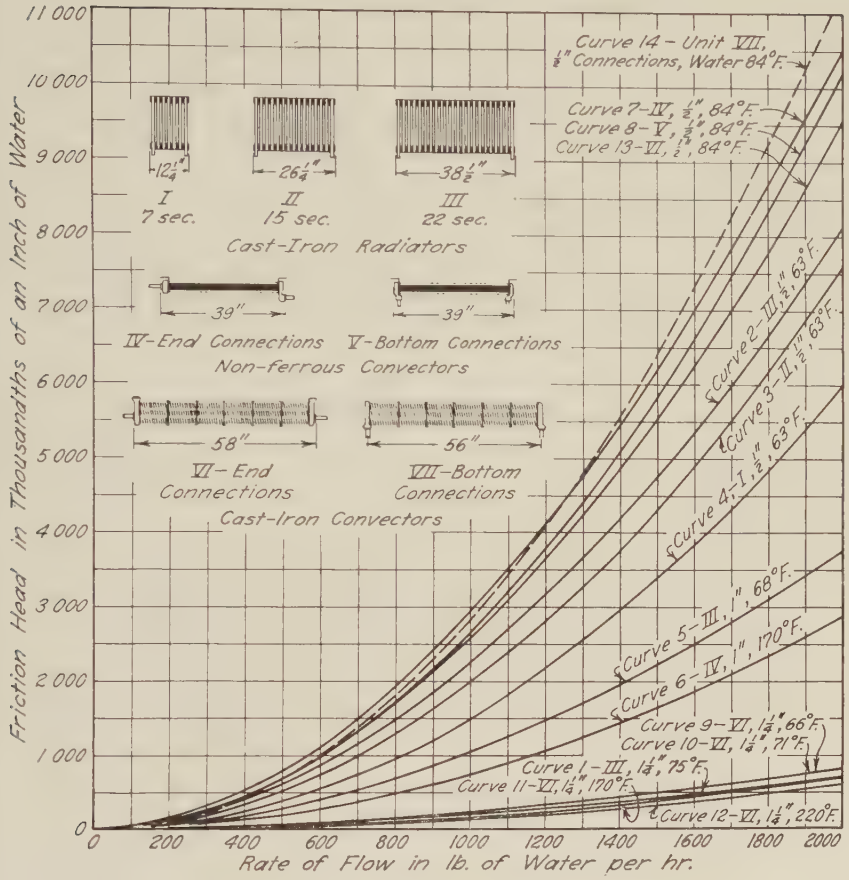

Fig. 29. Friction Heads of Various Radiators and Convectors

28. *Friction Head of 19-in., 4-tube, 22-section Radiator With 1½-in. Connections; Water Temperature 63 deg. F.*—The friction heads at a water temperature of 63 deg. F. and at various velocities were determined as described in Section 10 for the radiator with 1¼-in. connections and the following equation was derived:

$$H_f = 0.00589P^{1.86} \tag{22}$$

For this test, the temperature of the water was 63 deg. F. instead of 75 deg. F. because the temperature of the laboratory air was about 63 deg. F., and it was desired to have the temperature of the water practically the same as that of the air, so that the temperature of the water in the system might remain constant throughout the test.

The friction heads for the radiator, with ½-in. connections, calculated from Equation (22) are shown by curve 2 in Fig. 29. As indicated by the curve, the friction head in the radiator was 4750 mi. with the water flowing through the radiator at a rate of 1500 lb. per hr.; that is, with a water velocity of about 3.2 ft. per sec. in the ½-in. pipe connections.

29. *Variation in Friction Head With Length of Radiator.*—In this study, the 22-section radiator (Figs. 9, 10, 11, 15, and 16) was shortened successively to a 15-section and to a 7-section radiator. It was found that the friction heads for the 15-section and the 7-section radiators could be expressed by Equations (23) and (24) respectively.

$$H_f = 0.00299P^{1.94} \tag{23}$$

$$H_f = 0.00149P^{2.00} \tag{24}$$

The respective friction heads of the two radiators, calculated from Equations (23) and (24), are shown by curves 3 and 4 in Fig. 29.

It appears from curves 2, 3, and 4 in Fig. 29 that the friction head of a radiator decreases as the length of the radiator is decreased. The friction head of a radiator may therefore be divided into two parts—one part being the friction head at entrance and exit, and the other part, the friction head in the body of the radiator.

By a series of trial calculations of friction heads for small-tube radiators, it was found that for a 19-in., 4-tube radiator with ½-in. connections the relation between friction head, rate of water flow, and number of sections could be expressed, approximately, by the following equation:

$$H_f = 0.000395P^{2.18} + 0.001125P^{1.54}S \tag{25}$$

in which

H_f = friction head in mi. of water

P = rate of flow in lb. of water per hr.

S = number of sections in radiator

30. *Effect of Size of Push Nipples on Friction Head of Radiators.*—Since it had been found (Section 14) that there might be a variation as large as 8 per cent in the heat emission of radiators normally of

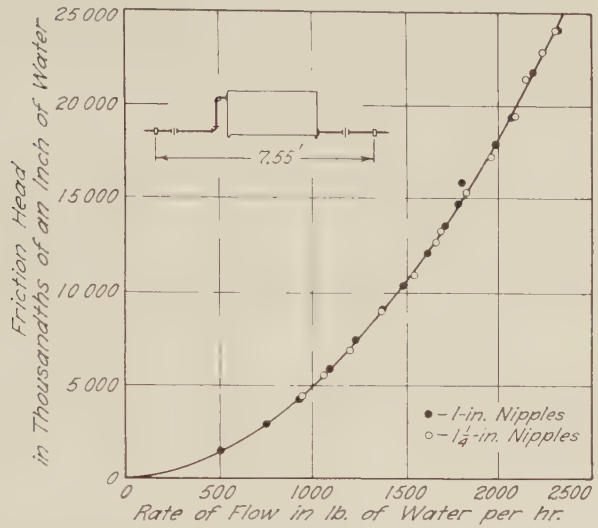

FIG. 30. COMPARISON OF FRICTION HEADS OF RADIATORS, INCLUDING PIPING, WITH 1¼-IN. AND 1-IN. PUSH NIPPLES

the same size but differing from each other slightly in details of construction, it was decided to determine the difference, if any, between the friction heads of two 19-in., 4-tube, 22-section radiators, one having 1-in. push nipples and the other, 1¼-in. push nipples.

The results of these tests, shown in Fig. 30 represent the friction heads for the radiator plus the two pipes connecting the radiator to the piezometer rings.

Since it is evident that there was no difference in the two sets of friction heads for the complete installations, it is safe to conclude that the friction head of a radiator is not changed appreciably by substituting 1¼-in. push nipples for 1-in. push nipples. This conclusion seems reasonable, since the larger part of the friction head of a radiator is in the inlet and outlet connections. Furthermore, the remaining part of the friction head, which is in the body of the radiator, occurs principally at the inlets and outlets of the push nipples, and only to a negligible extent within the push nipples themselves.

VII. FRICTION HEADS OF CONVECTORS

31. *Friction Head of Non-ferrous Convector With End Inlet and 1-in. Connections.*—For this study, the convector was installed as shown in Fig. 4, and tests were run at two water temperatures,

namely 68 deg. F. and 170 deg. F., using the method described in Section 11. It was found that the friction heads for the two water temperatures, 68 deg. F. and 170 deg. F., could be expressed by Equations (26) and (27) respectively.

$$H_f = 0.00293P^{1.85} \tag{26}$$

$$H_f = 0.00113P^{1.94} \tag{27}$$

The friction heads, calculated from Equations (26) and (27), are shown by curves 5 and 6 in Fig. 29. It appears from these two curves that, for a rate of flow of 1000 lb. per hr., the two friction heads are about 1030 and 750 mi., respectively, and that, for this rate of flow, the friction head for 170 deg. F. water is about 27 per cent lower than that for 68-deg. F. water.

32. *Friction Head of Non-ferrous Convector With End Inlet and ½-in. Connections.*—For this study, the convector was installed as shown in Fig. 4, and tests were run with a water temperature of 84 deg. F., following the method outlined in Section 11.

In this case the friction heads were expressed by the following equation:

$$H_f = 0.00701P^{1.87} \tag{28}$$

The friction heads, calculated from Equation (28), are shown by curve 7 in Fig. 29. It appears from Equations (26) and (28) and the corresponding curves in Fig. 29 that, for a flow of 1000 lb. per hr., the friction head of the convector was about 2.8 times as large for ½-in. connections as it was for 1-in. connections, if the small effect on the friction head resulting from the variation in water temperature of from 84 deg. F. to 68 deg. F. is neglected.

33. *Difference Between Friction Heads of Non-ferrous Convectors With End Connections and With Bottom Connections.*—For this study, the convector shown in Fig. 4 was installed as shown in Fig. 5. The same pipe sections and pipe fittings were used in the installation shown in Fig. 5 as had been used in the installation shown in Fig. 4, so that any difference in the friction heads of the two installations would result entirely from the difference between the two paths followed by the water in entering the convector.

For the convector with the bottom connection the equation for the friction head was found to be

$$H_f = 0.00372P^{1.95} \tag{29}$$

The friction heads, calculated from Equation (29), are shown by curve 8 in Fig. 29. A comparison of curves 7 and 8 shows that, at all water velocities, the end connection produced higher friction heads than those resulting from the bottom connection. However, the difference between the two is not large.

34. *Friction Head of Cast-iron Convector With End Inlet and Outlet and 1¼-in. Connections.*—The convector was installed as shown in Fig. 2, except that an end inlet and outlet were used, and was tested with 66 deg. F. water. The equation for friction head was found to be

$$H_f = 0.00047P^{1.89} \tag{30}$$

The friction head, calculated from Equation (30), is shown by curve 9 in Fig. 29.

35. *Variation in Friction Head of Cast-iron Convector With Variation in Temperature of Water.*—For this study, the installation shown in Fig. 2 was used, with end inlet and outlet instead of bottom inlet and outlet, and tests were made with water at temperatures of 66 deg. F., 71 deg. F., 170 deg. F., and 220 deg. F. During the tests with hot water, the convector was insulated so as to reduce the drop in temperature of the water within the convector as much as possible.

The equations for the respective friction heads at the four water temperatures were found to be

$$H_f = 0.00047P^{1.89} \tag{31}$$

$$H_f = 0.00046P^{1.89} \tag{32}$$

$$H_f = 0.000101P^{2.07} \tag{33}$$

$$H_f = 0.0000145P^{2.31} \tag{34}$$

The friction heads calculated by means of these four equations are shown by curves 9, 10, 11, and 12 in Fig. 29. It appears from these curves that the friction head decreases as the temperature of the water

is increased. However, the friction head in all four cases was so small that it is impossible to make accurate numerical comparisons.

36. *Difference Between Friction Heads of Cast-iron Convector With ½-in. End Connections and Those of Same Convector With ½-in. Bottom Connections.*—For this study, the installation shown in Fig. 2 was used for the tests with bottom connections. For the tests with end connections, the same pipe and pipe fittings were used but in a different sequence, as shown for the non-ferrous convector tests in Figs. 4 and 5, so that any difference in the friction heads of the two installations would result entirely from the difference between the two paths followed by the water in entering the convector.

The equations for the friction heads for the end and bottom connections respectively were found to be

$$H_f = 0.00553P^{1.89} \tag{35}$$

and

$$H_f = 0.00226P^{2.03} \tag{36}$$

The friction heads calculated from these equations are shown by curves 13 and 14 in Fig. 29. It appears from these curves that the friction heads for the end connections were lower at all except very low water velocities than the corresponding heads for the bottom connection. However, the differences between the two can be disregarded as far as design calculations are concerned.

37. *Friction Heads of Cast-iron Convector With 1¼-in. and ½-in. End Pipe Connections.*—A comparison of the relative friction heads at comparable water temperatures, produced by using end connections involving 1¼-in. and ½-in. pipe may be made from Equations (32) and (35), or from the corresponding curves 10 and 13 in Fig. 29. From the curves it appears that, at a rate of flow of 1000 lb. per hr., the friction head obtained with ½-in. pipe connections was about eleven times that obtained with 1¼-in. pipe connections.

38. *Friction Heads of Radiators and Convectors Expressed in Terms of Velocity Heads or Elbow Equivalents.*—With the velocity, V, expressed in ft. per sec., the velocity head in ft. of water is equal to $V^2/2g$, or to $0.0155V^2$. In expressing the friction head of a radiator in terms of velocity heads, the velocity referred to is that of the water in the pipe connected to the radiator. Since the friction head of one

TABLE 6
FRICTION HEADS OF RADIATORS AND CONVECTORS EXPRESSED IN VELOCITY HEADS, OR ELBOW EQUIVALENTS

	Size of Connections in.	Temperature of Water deg. F.	Velocity Heads or Elbow Equivalents for Rates of Flow of		
			500 lb./hr.	1000 lb./hr.	1500 lb./hr.
Radiators					
19-in., 4-tube, 22-section	1¼	75	6.0	5.5	5.2
19-in., 4-tube, 22-section	½	63	3.0	2.7	2.6
19-in., 4-tube, 15-section	½	63	2.5	2.3	2.3
19-in., 4-tube, 7-section	½	63	1.8	1.9	1.9
Convectors					
Non-ferrous, End Inlet	1	68	11.1	10.0	9.4
Non-ferrous, End Inlet	1	170	8.4	8.1	7.9
Non-ferrous, End Inlet	½	84	3.7	3.4	3.2
Non-ferrous, Bottom Inlet	½	84	3.3	3.1	3.2
Cast-iron, End Inlet	½	84	3.3	3.2	3.0
Cast-iron, Bottom Inlet	½	84	3.2	3.3	3.4
Cast-iron, End Inlet	1¼	66	6.9	6.4	6.1
Cast-iron, End Inlet	1¼	71	6.6	6.3	6.0
Cast-iron, End Inlet	1¼	170	4.3	4.5	4.7
Cast-iron, End Inlet	1¼	220	2.6	3.3	3.7

90-degree standard screwed elbow is practically equal to $V^2/2g$, or to one velocity head, it is customary to express the friction heads of radiators in terms of the friction heads of 90-degree standard screwed elbows, otherwise designated as elbow equivalents.

For example, if the friction head of a 19-in., 4-tube, 22-section radiator with ½-in. connection is 2250 mi. with 63-deg. F. water flowing through the radiator at the rate of 1000 lb. per hr., the velocity of the water in the ½-in. pipe is 2.105 ft. per sec.; the velocity head is 0.0688 ft. of water or 826 mi. of water. The friction head of the radiator is therefore 2250/826 or 2.7 velocity heads, or, in other words, 2.7 elbow equivalents.

The friction heads for the different radiators and convectors tested, expressed in velocity heads, or in elbow equivalents, shown in Table 6 for rates of flow of 500, 1000, and 1500 lb. of water per hr., were calculated from the friction heads in milinches determined by means of the appropriate equations developed in Sections 27-36 inclusive.

VIII. CONCLUSIONS

39. *Conclusions.*—The following conclusions may be drawn as applying to the test data and the conditions under which the tests were conducted:

(1) The friction head produced by the water flowing through a radiator or convector may be divided into three parts, namely, the

friction head at inlet, the friction head at outlet, and the friction head within the unit itself. Of these three friction heads, the friction head at inlet is the greatest; the friction head at outlet is intermediate; and the friction head in the unit is the smallest except in the case of very long units. For the 4-tube radiators tested with ½-in. pipe connections, it was found that the sum of the friction heads at inlet and outlet varies as the 2.18 power of the rate of flow of the water, and that the friction head in the body of the radiator varies directly as the length of the radiator and directly as the 1.54 power of the rate of flow of the water. The formula developed for this case is

$$H_f = 0.000395P^{2.18} + 0.001125P^{1.54}S \qquad (37)$$

in which

H_f = friction head in milinches of water

P = rate of flow of water through the radiator in lb. per hr.

S = number of sections per radiator

(2) The sum of the friction heads at inlet and outlet in the case of convectors is practically the same as that for radiators, since there is practically no difference in the course of the water through these two types of units.

(3) The friction head produced within a radiator or convector is determined by the shape, size, and length of the passages through which the water flows.

(4) For a given rate of flow of water, the size of the connection has an important influence on the friction heads at inlet and outlet and, therefore, also on the overall friction head. For example, using the friction head for a 1¼-in. connection as the basis, the friction head for a 1-in. connection would be about four times as large, and that for a ½-in. connection about twelve times as large.

These large differences may be explained as follows. If a ½-in. pipe and a 1-in. pipe are to supply water to a radiator or convector at the same rate, the velocity of the water in the ½-in. pipe must be 2.84 times the velocity in the 1-in. pipe; the kinetic energy of the water in the ½-in. pipe will then be eight times that of the water in the 1-in. pipe. As the water enters the radiator or convector, its velocity is reduced and the kinetic energy of the stream of water is transformed partly into pressure head and partly into heat. The latter portion constitutes a total loss in so far as the pressure head produced by the flowing stream of water is concerned.

(5) The differences between the friction heads for end and bottom connections of the two convectors tested were so small that they may be considered negligible. In both cases, the friction heads for bottom connections were slightly smaller at low water velocities and slightly larger at high water velocities.

(6) The effect of an increase in the temperature of the water is to reduce the friction head in the radiator or convector.

In the case of the cast-iron convector it was found that increasing the water temperature from 66 deg. F. to 220 deg. F. resulted in a reduction of about 40 per cent in the friction head.

In the case of the non-ferrous convector it was found that increasing the water temperature from 68 deg. F. to 170 deg. F. resulted in a reduction of about 14 per cent in the friction head.

(7) For the purpose of design, it is best to express the friction head of a radiator or convector in terms of elbow equivalents or velocity heads, as defined in Section 38.

The value of the friction head of a radiator, expressed in elbow equivalents or velocity heads, ranges from about two for a short radiator with ½-in. connections and a high velocity of the water to about six for a long radiator with 1¼-in. connections and a low velocity of the water.

(8) The friction head of a *long* cast-iron convector, expressed in elbow equivalents or velocity heads, varies from about three for a convector with ½-in. connections and a high velocity of the water, to about six for a convector with 1¼-in. connections and a low velocity of the water.

(9) The friction head of a *long* non-ferrous convector, expressed in elbow equivalents or velocity heads, varies from about three for a convector with ½-in. connections and a high velocity of the water, to about eleven for a convector with 1-in. connections and a low velocity of the water.

(10) For convectors which are shorter than those included in these tests, the friction heads are, no doubt, smaller than those listed in Table 6.

(11) For the majority of the cases occurring in practice, it is sufficiently accurate to assume the friction head of a radiator or convector as equal to that of three standard 90-degree screwed elbows, or to three elbow equivalents. If the water velocity, V, is expressed in inches per second, the friction head of one radiator or convector is $3.88V^2$ milinches of water.

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